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PRELIMINARY DETERMINATIONS OF TEMPERATURE LIMITATIONS OF ESTER, ETHER, AND HYDROCARBON BASE LUBRICANTS IN 25-MM BORE BALL BEARINGS

by Erwin V. Zaretsky and William J. Anderson

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Page 13: In table IV, the multiplication factors for the Estimated maximum Hertz stress should be 10^3 instead of 10^6.

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SUMMARY

Groups of 7205-size (25-mm bore) angular-contact ball bearings made from AISI M-1 steel were tested with eleven high-temperature lubricants. These lubricants were classified as polyphenyl ethers, esters, and hydrocarbons. Test conditions included outer-race temperatures of 250° to 600° F (394° to 588° K), speeds from 20 000 to 45 000 rpm, and maximum Hertz stresses of 189 000 to 347 000 psi (130 000 to 239 000 N/cm^2) in an SKF high-temperature bearing tester.

Bearing AFBMA-rated (catalog) life was exceeded at a temperature between 550° and 600° F (560° and 588° K) with a synthetic paraffinic oil with an anti-wear additive under a low oxygen environment. The estimated bearing life with a diester base lubricant was less than half bearing AFBMA-rated (catalog) life in the temperature range of 450° to 550° F (505° to 560° K).

Of the three classes of lubricants tested, the polyphenyl ethers exhibited the poorest life potential. Early fatigue spalling, wear, and surface distress occurred in the rolling-element components. The bearings run with the polyphenyl ethers did have the ability to operate in an air environment. However, sludge formations occurred with this type lubricant in both air and limited oxygen environments.

Heavy coke deposits formed on bearings run with the ester lubricants at a temperature of 500° F (533° K) in a low oxygen environment. Gross lubricant degradation did not occur with the hydrocarbon lubricants run at 600° F (588° K) in the low oxygen environment, although the fluid became blackened and some sludge deposits were noticed.

INTRODUCTION

When the advent of gas turbine powerplants first pushed aircraft engine bearing operating temperatures beyond the capabilities of the mineral oils, the development of synthetic lubricating fluids was undertaken. The guiding philosophy of most lubricant development was to obtain high-temperature thermal and oxidative stability in fluids having relatively limited viscosity variation over wide temperature ranges while at the same time reducing lubricant volatility (ref. 1). The most notable results of this early research were the diester base lubricants which have been used extensively and successfully in aircraft jet engines. The military designation and specification for this lubricant is MIL-L-7808. This lubricant combines the thermal stability of the ester grouping in a large molecule with a low variation of viscosity with temperature that is characteristic of the paraffinic-type constituents in this fluid. The lubricant is capable of being pumped at -65° F (220° K) and, with a suitable additive package, is capable of operation at a bulk-oil temperature in a jet engine of 300° F (422° K). Additive refinements and other ester-base lubricants with higher viscosities than the MIL-L-7808 specification lubricants have increased the upper bulk-oil temperature limits to approximately 400° F (478° K).

Recent advances in synthetic lubricant development have been made with types having higher inherent thermal and oxidative stability than the paraffinic straight-chain configurations such as in the di-2-ethylhexyl sebacate. Such a lubricant is a polyphenyl ether (refs. 2 and 3) which has considerable radiation stability but poor viscosity-temperature characteristics. Thus, the polyphenyl ethers will generally have high pour points when compared with the more conventional lubricating fluids.

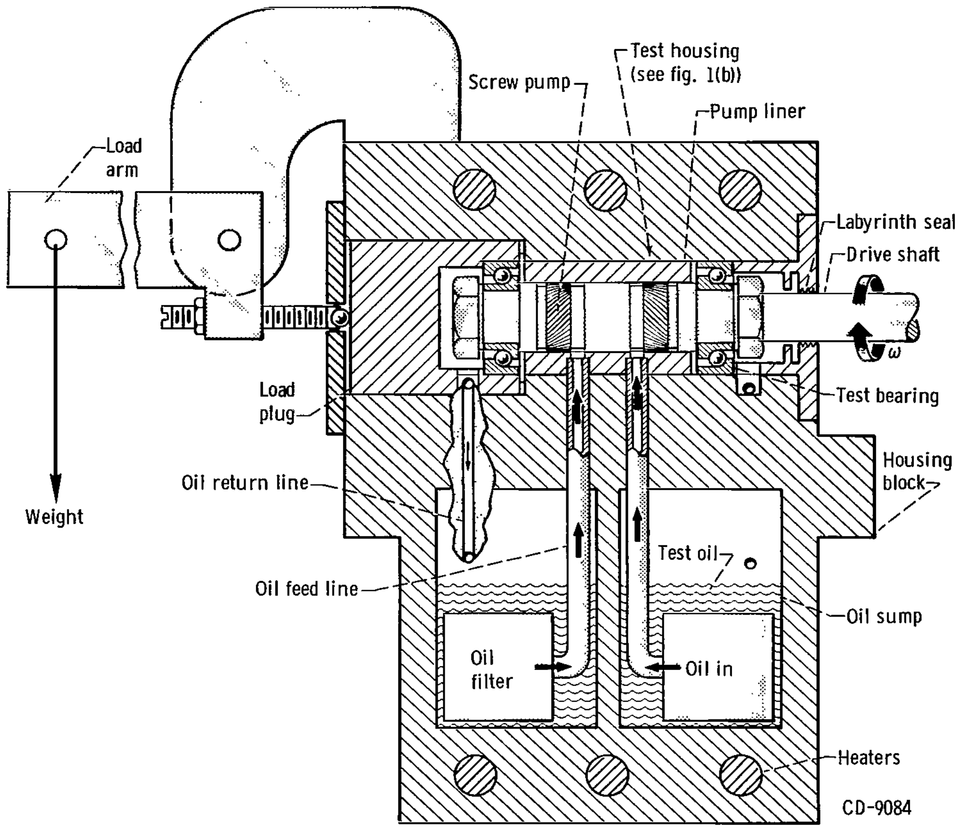
Other classes of fluids which have shown superior high-temperature characteristics are the synthetic paraffinics and super-refined mineral oils (refs. 4 and 5). These fluids can have useful liquid ranges from -20° to 700° F (245° to 644° K). In addition, they will accept additives which improve their boundary lubricating characteristics.

The research reported herein was undertaken to evaluate the performance and potential of the best available high-temperature lubricants in a bearing under extreme environmental conditions. The objectives were the following:

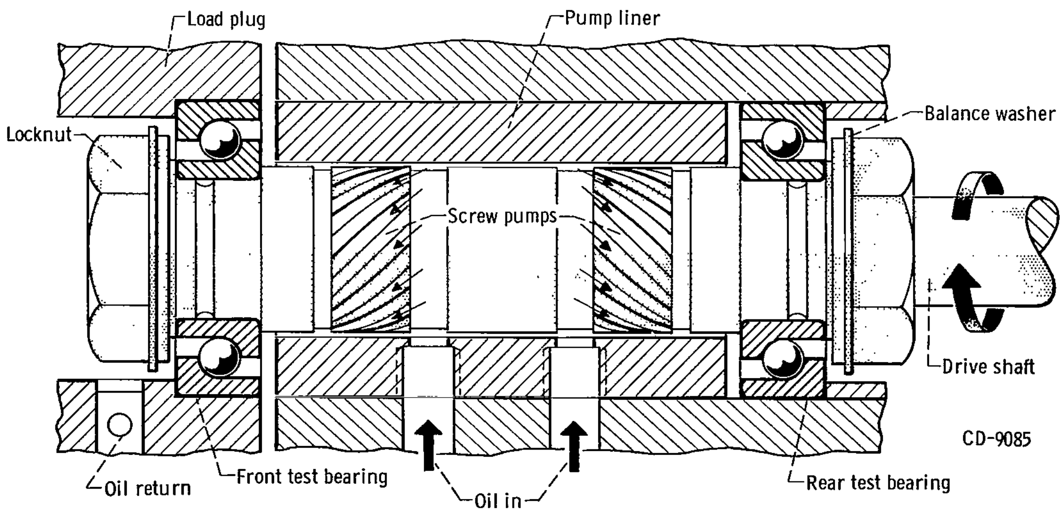
(1) To investigate, with 7205-size (25-mm bore) angular-contact ball bearings, the limiting load and temperature for long-term reliable operation of high-temperature lubricants; and

(2) To determine the endurance characteristics of two groups of 7205-size angular-contact ball bearings run with the two most favorable lubricants tested.

Tests of eleven lubricants having varied chemical and physical characteristics were conducted at temperatures to 600° F (588° K) with 7205-size angular-contact ball bearings made of vacuum-melted M-1 steel having a Rockwell C hot hardness in excess of 60 at operating temperature. Test conditions included speeds from 20 000 to 45 000 rpm and bearing thrust loads from 150 to 918 pounds (668 to 4090 N) which produced maximum Hertz stresses ranging from 189 000 to 347 000 psi (130 000 to 239 000 N/cm^2) on the bearing inner race. Based upon the appearance of the bearing (i.e., the surface appearance of the races and balls), lubricant deposit formations, and temperature rise in the bearing during operation, the bearing life potential was estimated with respect to lubricant type and viscosity. All tests were conducted in a SKF high-temperature bearing fatigue tester by SKF Industries, Inc. under contract to NASA and initially reported in reference 6.



(a) Schematic of tester.



(b) Test housing.

Figure 1. - SKF high-speed, high-temperature bearing test apparatus.

TEST APPARATUS

A high-speed, high-temperature bearing test apparatus developed by SKF Industries, Inc. (ref. 6) was used for the tests reported herein. A schematic of this apparatus is shown in figure 1(a). The apparatus in essence comprises a test housing, shown in figure 1(b), and a drive shaft which has the test bearings fastened to it by lock nuts. Integral with the test shaft are two screw pumps. In operation the lubricant from an oil sump flows through oil feed lines to the screw pumps which pump the lubricant to the test bearings. The oil then returns by means of gravity to the oil sump. The oil sump and the test housing are inerted by nitrogen gas. Cartridge-type heaters maintain temperature in the test housing and oil sump. Loading of the bearings is achieved by a deadweight system which applies a force against a load plug which thrust loads the front and rear test bearings (fig. 1(a)). Drive is supplied to the drive shaft through a belt drive and a speed increaser gear box (not shown in fig. 1) capable of providing shaft speeds up to 50 000 rpm. Additionally, the apparatus is capable of temperatures to 1000° F (811° K) and bearing thrust loads to 1000 pounds (4450 N).

Temperatures are monitored by thermocouples on the two test bearing outer races, in the oil sump, the rig housing block above and below the test housing, and the pump liner. The oil sump temperature can be controlled within $\pm 13^{\circ}$ F ($\pm 7.2^{\circ}$ K). The sump bulk temperature can be maintained no less than 60° F (33° K) lower than that of the test bearing temperature at a speed of 42 800 rpm. Instrumentation provides for temperature control and automatic failure detection because of increased temperature or vibration. The mean temperatures of the two test bearings are not controlled separately. Experience has shown that the mean bearing temperatures in any specific test usually differ by less than 17° F (9.5° K).

TEST SPECIMENS AND MATERIAL

Test Bearings

Test bearings were manufactured to a 7205-size angular-contact design having counterbored outer races and inner-race riding cages (separators) and finished to an ABEC 5 specification. All inner and outer races of the test bearing were made from the same heat of material. The balls were made from a separate heat of material. For purposes of initial testing, some bearings were manufactured with a nominal contact angle of 17°, a radial looseness of 30 to 40 microns, and conventional narrow-width cages. However, most testing was performed with bearings having a nominal contact angle of 19°, a radial looseness of 40 to 50 microns, and a wide-land, silver-plated, hardened cage made

TABLE I. - CONTACT STRESSES FOR 7205-SIZE ANGULAR-CONTACT BALL BEARING WITH CONFORMITIES ON INNER AND OUTER RACES OF 52.2 AND 53.2 PERCENT, RESPECTIVELY

Contact angle, deg	Speed, rpm	Thrust load		Maximum Hertz stress			
				Inner race		Outer race	
		lb	N	psi	N/cm^2	psi	N/cm^2
17	20 000	150	667	189×10^3	130×10^3	183×10^3	176×10^3
		295	1315	243	167	229	158
		365	1625	268	184	249	171
		450	2000	279	192	261	180
		580	2580	302	208	281	193
		725	3230	323	222	300	207
19	20 000	150	667	190×10^3	131×10^3	183×10^3	126×10^3
		365	1625	259	178	242	167
		725	3230	309	213	288	198
		918	4090	334	230	310	214
	42 800	365	1625	220×10^3	151×10^3	233×10^3	160×10^3
		459	2040	244	168	250	172
	45 000	365	1625	217×10^3	149×10^3	233×10^3	160×10^3

of AISI M-1 steel of nominal (room temperature) hardness Rockwell C 58. The inner and outer races of the latter groups of bearings were black oxide coated to improve their resistance to surface distress because of marginal lubricating conditions. The outer-race groove surfaces were finished to 6 to 8 microinches (0.152 to 0.203 μm) rms and the inner-race groove surface to 2 to 4 microinches (0.051 to 0.102 μm) rms. The variations of maximum Hertz stress as a function of speed and load are given in table I for the bearings.

The bearings were manufactured from consumable electrode vacuum-melted M-1 bearing steel. The nominal chemical composition of this material by percent weight is as follows:

Element	Percent weight
Carbon	0.80
Manganese	.30
Phosphorus	.030 (max)
Sulphur	.030 (max)
Silicon	.30
Chrome	4.00
Vanadium	1.00
Tungsten	1.50
Molybdenum	8.00
Iron	Remainder

This material was chosen because of its practical interest and its ability to maintain hot hardness at 600° F (588° K). Representative curves of hardness against temperature for M-1 and SAE 52100 steels are shown in figure 2.

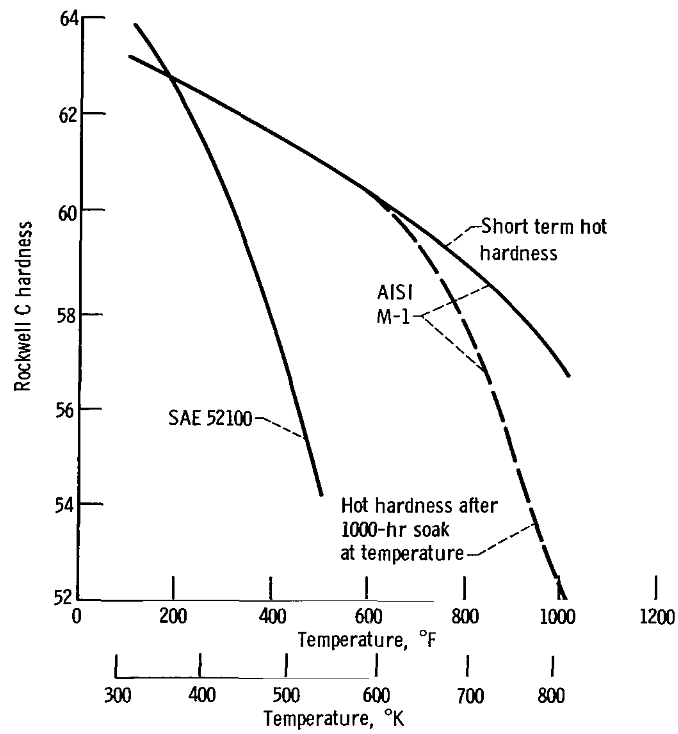


Figure 2. - Representative hot hardness as function of temperature for two bearing steels.

Test Lubricants

Eleven lubricants which were considered of current interest were selected for study. Each lubricant varied either with respect to base stock, viscosity at atmospheric conditions, or additive content. These eleven lubricants can be classified as three basic types: (1) polyphenyl ethers, (2) esters, and (3) hydrocarbons. A summary of the properties of these lubricants can be found in table II. Figure 3 contains ASTM standard viscosity-temperature charts based on manufacturers' data for these lubricants. These lubricants were not outgassed before testing.

TABLE II. - TEST LUBRICANT PROPERTIES

Lubricant type	Lubricant designation	Base stock	Additive content	Representative viscosity, cs[a]		
				100° F (311° F)	210° F (372° K)	[b]500° F (533° K)
Modified polyphenyl ether	A	Blend of 3-ring and 4-ring components	Not available	26	4.3	0.82
	B	Blend of 3-ring and 4-ring components	Not available	56	5.9	.85
Polyphenyl ether	C	5P4E	None	365	13.1	1.07
	D	5P4E	(c)	365	13.1	1.07
	E	6P5E	None	1831	24.7	1.20
Ester	F	Mixed polyester-diester	Not available	40	8.4	1.70
	G	Diester	(c), (d), (e)	37	7.8	1.50
Hydrocarbon	H	Synthetic paraffinic	None	314	32	2.9
	I	Synthetic paraffinic	(d)	314	32	2.9
	J	Super-refined naphthenic mineral oil	(c), (d), (e)	79	8.4	1.1
	K	Super-refined paraffinic mineral oil	None	480	28	2.1

[a]Manufacturers' data.
[b]Estimated.
[c]Oxidation inhibitor.
[d]Anti-wear additive.
[e]Anti-foam agent.

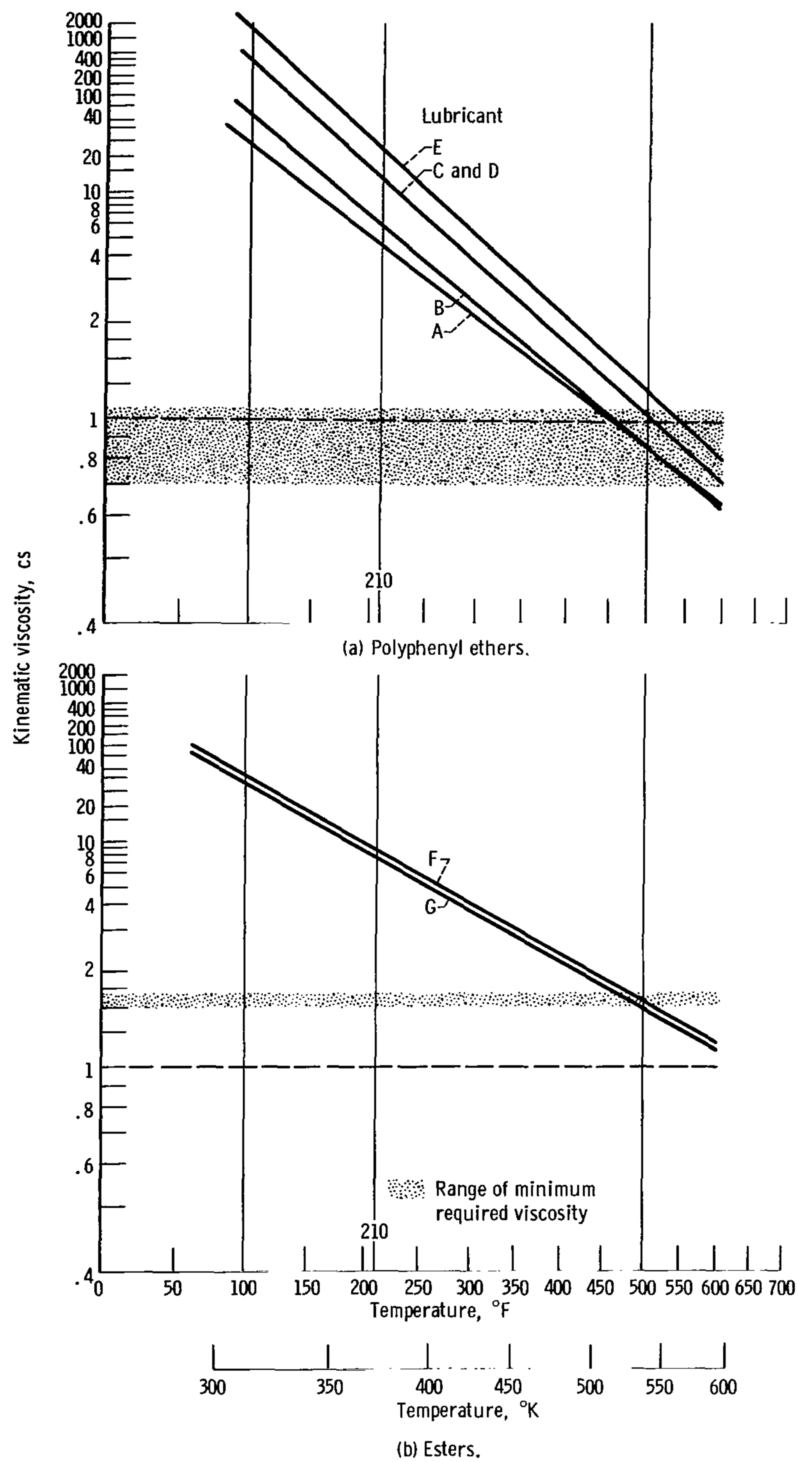
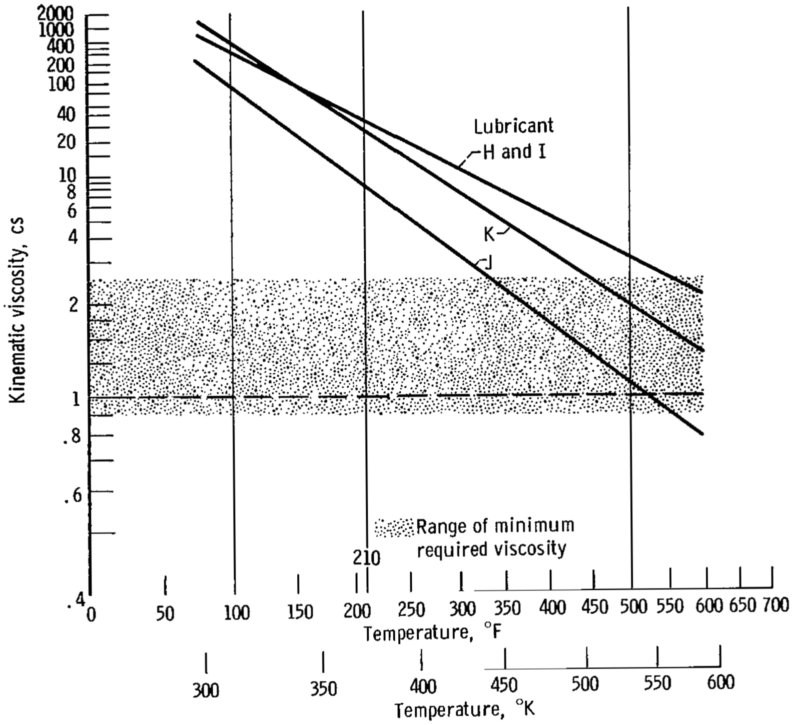


Figure 3. - Representative viscosity of experimental lubricants as function of temperature.



(c) Hydrocarbons.

Figure 3. - Concluded.

TEST PROCEDURE

The bearings were assembled in pairs in the test apparatus and approximately 2 quarts of test lubricant placed in the oil sump (see fig. 1(a)). Nitrogen was introduced to pressurize the oil sump and provide a low oxygen environment. The lubricant was thus forced through the oil filter to prime the screw pump. The oil and the bearings were subsequently heated to test temperature. Testing was conducted at drive shaft speeds from 20 000 to 45 000 rpm.

Lubricant flow to each bearing was maintained in a range of 90 to 390 cubic centimeters per minute depending upon the viscosity of the lubricant at test temperature. The mean test temperature of the outer race was varied for some of the lubricants from 250° F (394° K) to approximately 650° F (616° K). The bearing thrust load was also varied from 150 to 918 pounds (668 to 4090 N).

RESULTS AND DISCUSSION

Groups of 7205-size (25-mm bore) angular-contact ball bearings made from consumable-electrode vacuum-melted AISI M-1 steel were tested with the eleven lubricants. These lubricants varied either with respect to base stock, additive content, or viscosity at room temperature. Test conditions included speeds from 20 000 to 45 000 rpm and bearing thrust loads from 150 to 918 pounds (668 to 4090 N) producing maximum Hertz stresses ranging from 189 000 to 347 000 psi (130 000 to 239 000 N/cm^2) on the bearing inner race. The test conditions and results are summarized in tables III(a) to (k).

Polyphenyl Ethers

Three polyphenyl ether base stocks were investigated. These lubricants were a 6P5E, a 5P4E, and modified polyphenyl ether which is a blend of 3-ring and 4-ring compounds. The results with these lubricants are summarized in table III(a) to (e) and table IV.

Lubricant designations A and B were the modified polyphenyl ethers which differed with respect to room temperature viscosity. At the elevated temperatures, that is, 500° F (533° K) (see fig. 3(a)), viscosity differences were negligible and were not expected to have any effect on bearing operation. Accordingly, bearings run with lubricants A and B indicated a maximum operating capability of over 550° F (560° K) and a contact stress of over 250 000 psi (172 000 N/cm^2).

Similar results were obtained with lubricant C, the 5P4E base stock without an additive. As expected, the addition of an oxidation inhibitor to the 5P4E base stock, lubricant D, did not appear to increase bearing life with this base stock in a low oxygen environment.

Operation in an air environment with lubricant D resulted in an apparent increase in life and improved bearing race appearance. However, the improvement in life was not considered significant. The results indicated that the 5P4E polyphenyl ether with an oxidation inhibitor has high-temperature potential beyond 600° F (588° K) in an oxidative environment. These results are in accordance with tests reported in reference 7 for the same lubricant base stock having a similar oxidation inhibitor.

Results obtained with the 6P5E base stock without an inhibitor indicate that this fluid may have a higher temperature potential (i.e., greater than 650° F (616° K)) than the previous two base stocks discussed. However, testing with this fluid was limited because of problems with its availability.

Based upon the maximum temperature at which the polyphenyl ethers were able to lubricate the bearings, the range of required minimum viscosity was estimated from the

TABLE III. - SUMMARY OF TEST CONDITIONS AND RESULTS FOR TEST LUBRICANTS IN A LOW OXYGEN ENVIRONMENT

Speed, rpm	Thrust load, lb	Thrust load, N	Temperature range, °F	Temperature range, °K	Bearings tested: Total number	Bearings tested: Number failed[a]	Bearings tested: Number fatigued	Bearings tested: Number unfailed	Maximum operating time for any single bearing, hr
(a) Lubricant A - modified polyphenyl ether, blend of 3-ring and 4-ring components; bearing contact angle, 19°									
42 800	365	1625	500 - 550	533 - 560	2	1	--	1	23
(b) Lubricant B - modified polyphenyl ether, blend of 3-ring and 4-ring components; bearing contact angle, 19°									
42 800	459	2040	350 - 400	450 - 478	2	1	--	1	<1
	459	2040	550 - 600	560 - 588	2	--	1	1	30
(c) Lubricant C - 5P4E polyphenyl ether; bearing contact angle, 17°									
20 000	365	1625	500 - 550	533 - 560	2	--	--	2	2
	365	1625	550 - 600	560 - 588	2	--	2	--	2
	450	2000	550 - 600	560 - 588	2	--	1	1	1
	580	2580	600 - 650	588 - 616	2	--	2	--	2
	725	3230	550 - 600	560 - 588	[b]4	--	4	--	4
	918	4090	550 - 600	560 - 588	[b]2	--	--	2	5
	918	4090	600 - 650	588 - 616	[b]2	--	2	--	5
(d) Lubricant D - 5P4E polyphenyl ether; bearing contact angle, 19°									
42 800	365	1625	250 - 300	394 - 422	6	4	--	2	<1
	459	2040	250 - 300	394 - 422	12	6	--	6	<1
	↓	↓	400 - 450	478 - 505	3	--	--	3	<1
	↓	↓	450 - 500	505 - 533	2	--	1	1	1
	↓	↓	500 - 550	533 - 560	6	1	2	3	2
	↓	↓	550 - 600	560 - 588	3	--	2	1	38
	↓	↓	[c]550 - 600	560 - 588	4	1	1	2	78
(e) Lubricant E - 6P5E polyphenyl ether; bearing contact angle, 19°									
45 000	365	1625	550 - 600	560 - 588	1	--	--	1	1
	365	1628	600 - 650	588 - 616	1	--	1	--	1
42 800	365	1628	600 - 650	588 - 616	2	--	--	2	20
(f) Lubricant F - mixed polyester diester; bearing contact angle, 19°									
42 800	365	1625	500 - 550	533 - 560	4	3	--	1	90

TABLE III. - Concluded. SUMMARY OF TEST CONDITIONS AND RESULTS FOR TEST LUBRICANTS IN A LOW OXYGEN ENVIRONMENT

Speed, rpm	Thrust load, lb	Thrust load, N	Temperature range, $^{\circ}$F	Temperature range, $^{\circ}$K	Bearings tested: Total number	Number failed[a]	Number fatigued	Number unfailed	Maximum operating time for any single bearing, hr
(g) Lubricant G - diester; bearing contact angle, 19°									
42 800	365	1625	450 - 500	505 - 533	14	2	--	12	90
	365	1625	500 - 550	533 - 560	16	2	1	13	90
	459	2040	450 - 500	505 - 533	14	--	4	10	90
	459	2040	500 - 550	533 - 560	16	4	5	7	90
(h) Lubricant H - synthetic paraffinic oil; bearing contact angle, 19°									
42 800	459	2040	450 - 500	505 - 533	3	1	--	2	1
	459	2040	500 - 550	533 - 560	9	2	3	4	<10
	459	2040	550 - 600	560 - 588	6	--	--	6	>90
(i) Lubricant I - synthetic paraffinic oil; bearing contact angle, 19°									
42 800	459	2040	550 - 600	560 - 588	10	--	--	10	>280
(j) Lubricant J - super-refined naphthenic mineral oil; bearing contact angle, 17°									
20 000	150	667	250 - 300	394 - 422	2	--	--	2	1
			400 - 450	478 - 505	[d]2	1	--	1	8
			450 - 500	505 - 533	4	1	--	3	75
			500 - 550	533 - 560	[d]6	1	1	4	75
			550 - 600	560 - 588	4	1	--	3	35
			600 - 650	588 - 616	4	2	--	2	15
	365	1625	500 - 550	533 - 560	[b]2	--	2	--	30
	365	1625	550 - 600	560 - 588	2	--	2	--	15
	450	2000	400 - 450	478 - 505	2	1	--	1	>1
(k) Lubricant K - super-refined paraffinic mineral oil; bearing contact angle, 19°									
20 000	295	1315	550 - 600	560 - 588	[e]2	--	1	1	30
42 800	459	2040	250 - 300	394 - 422	5	2	--	3	<1
			350 - 400	450 - 478	2	2	--	--	<1
			400 - 450	478 - 505	1	--	--	1	<1
			450 - 500	505 - 533	1	--	--	1	>90
			500 - 550	533 - 560	1	--	--	1	>1
			550 - 600	560 - 578	6	3	1	2	90
45 000	365	1625	500 - 550	533 - 560	4	--	2	2	90

[a]Smearing is predominent failure mode.
[b]Bearings run with 19° contact angle.
[c]Air environment.
[d]One bearing run with 19° contact angle.
[e]Bearings run with 17° contact angle.

TABLE IV. - ESTIMATE OF LUBRICANT TEMPERATURE, STRESS, AND LIFE LIMITS BASED ON OPERATION OF 7205-SIZE ANGULAR-CONTACT BEARINGS IN A LOW OXYGEN ENVIRONMENT

Lubricant type	Lubricant designation	Estimated maximum bearing operating temperature		Estimated maximum Hertz stress		Estimated life potential at maximum temperature and stress, (a)	Apparent minimum viscosity for avoidance of ball-race contact surface damage, cs
		°F	°K	psi	N/cm^2		
Polyphenyl ether	A	>500	>533	>233×10^{3}	>160×10^{3}	Fair	0.7 - 1.0
	B	>550	>560	>250	>172	Fair	
	C	>600	>588	>302	>208	Poor	
	D	>600	>588	>250	>172	Fair[b]	
	E	>600	>588	>233	>160	Fair	
Ester	F	>500	>533	>233×10^{3}	>160×10^{3}	Good	1.5 - 1.7
	G	>500	>533	>250	>172	Good	
Hydrocarbon	H	>550	>560	>250×10^{3}	>172×10^{3}	Good	0.9 - 2.5
	I	>550	>560	>250	>172	Excellent	
	J	>550	>560	>268	>184	Good	
	K	>550	>560	>250	>172	Good	

[a]Comparison based on performance of lubricant I.
[b]Air environment.

viscosity-temperature graph of figure 3(a). The minimum required viscosity at operating temperature with this type base stock was estimated to be 0.7 to 1.0 centistokes. The advantage of this type of base stock appears to be its capability to function in an oxidative (air) environment.

Ester Base Lubricants

Two ester base lubricants were evaluated for their high-temperature capability; these were lubricants F and G, a mixed polyester diester and a diester base stock, respectively. The results with these two lubricants are presented in table III (f) and (g). The number of tests run with lubricant F was limited. However, the results with lubricant G were based on a total of 60 tests. Surface distress occurred at the 500° F (533° K) temperature level with both lubricants, but the surface damage was limited to a relatively small number of bearings and did not appear as severe as that for the polyphenyl ethers. The critical problem with these lubricants was the formation of hard coke deposits which limit practical bearing operating temperatures to less than 500° F (533° K). These deposits were present to some degree in all the bearings tested. Additionally, cage wear appeared to be somewhat of a problem but not a mode of failure.

The Anti-Friction Bearing Manufacturers Association (AFBMA) rated (catalog) life or the bearing 10-percent life for this size bearing, which would normally be made of SAE 52100 air-melted steel, thrust loaded to 459 and 359 pounds (2040 and 1600 N) are approximately 90 and 180 hours, respectively. The actual life of these bearings was less than half catalog expectations for lubricant G. No life estimate could be made for the bearings run with lubricant F because of the limited amount of data.

Based upon the limiting temperature of 500° F (533° K), a minimum viscosity requirement was determined from the viscosity-temperature graph of figure 3(b). It is estimated that a viscosity of approximately 1.5 to 1.7 centistokes is required under the conditions reported for the ester base lubricants. Generalization of these results to other conditions is dependent on elastohydrodynamic theory (ref. 8).

Hydrocarbon Base Lubricants

Tests were conducted with four hydrocarbon base lubricants (table II). Surface distress was observed to some degree in bearings run with three of the four lubricants

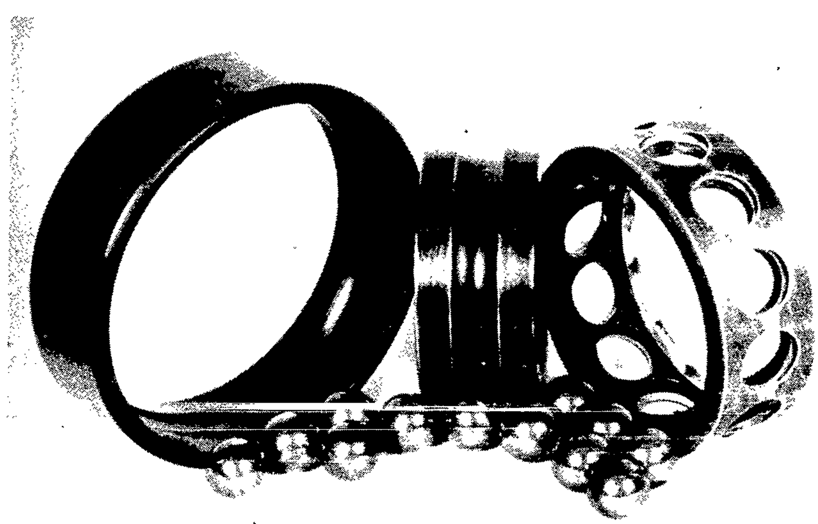


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Figure 4. - Bearing with lubricant I, synthetic paraffinic oil with an anti-wear additive after 283 hours of operation. Speed, 42 800 rpm; thrust load, 459 pounds (2040 N); temperature, 600° F (588° K) under low oxygen environment.

designated as lubricants H, J, and K (tables III (h), (j), and (k), respectively). Minimum surface distress occurred with lubricant H, the synthetic paraffinic, and lubricant K, the super-refined paraffinic. Lubricant I is the same as lubricant H, but with an anti-wear additive added (table III (i)). The results obtained with this lubricant were superior to all the other lubricants tested. All ten bearings tested at 42 800 rpm, a thrust load of 459 pounds (2040 N), and temperature range of 550° to 600° F (560° to 588° K) ran to over 190 hours before suspension. All contacting elements were in excellent condition subsequent to test, as can be seen in figure 4. No signs of any surface distress or wear were apparent. (The AFBMA-rated (catalog) life for this size bearing, which would normally be made of air-melted SAE 52100 steel, for a 459-pound (2040-N) thrust load is approximately 90 hr at 42 800 rpm.) These data indicate that not only can a bearing be run in the temperature range of 550° to 600° F (560° to 588° K), but lives exceeding catalog expectation can be achieved. Further, it is speculated that anti-wear additives added to the synthetic paraffinic oil may inhibit surface distress and extreme wear of the contacting elements. Based upon a temperature of 500° F (533° K), the apparent minimum viscosity from the temperature-viscosity graph of figure 3(c) for avoidance of surface damage is approximately 0.9 to 2.5 centistokes.

The apparent disadvantage of the hydrocarbon base fluids is that, for elevated temperatures, inerting of the lubrication system is required in order to prevent lubricant oxidation. This problem also occurs with the esters.

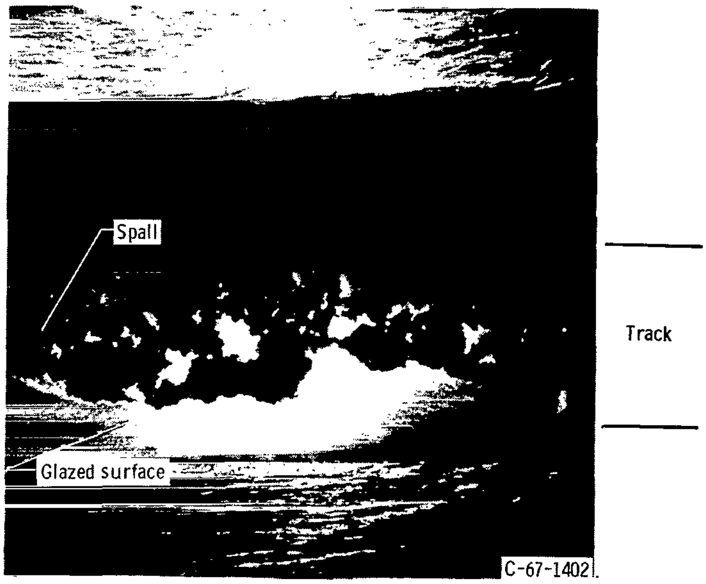
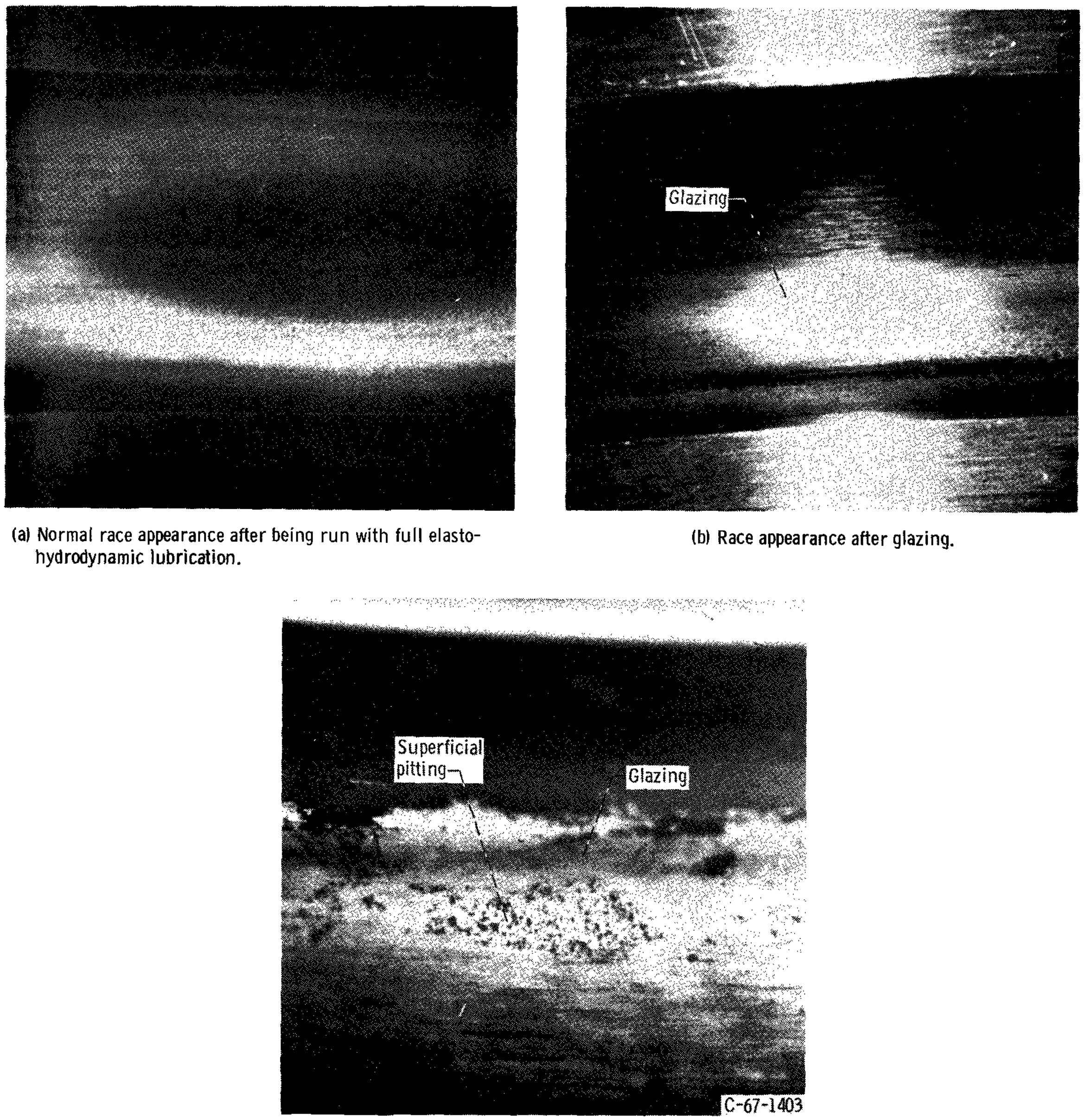


Figure 5. - Representative high-temperature fatigue spall.

Failure Modes

Types of high-temperature failure modes were discussed in references 8 and 9. These modes can be categorized as fatigue pitting, surface glazing and pitting, and surface smearing or deformation. The fatigue failures reported herein were generally more extensive and were associated with more surface distress (glazing and superficial pitting) than classical fatigue spalls normally experienced under more conventional temperature environments. An example of this fatigue failure is given in figure 5. A normal appear-



(a) Normal race appearance after being run with full elastohydrodynamic lubrication.

(b) Race appearance after glazing.

(c) Race appearance after glazing and superficial pitting.

Figure 6. - Race appearance representative of bearings run at elevated temperatures.

ing race is shown in figure 6(a) for comparison. As can be seen from figure 5 a certain amount of surface glazing is apparent. This condition was present in all bearings run with the various lubricants except lubricant I. The glazing was more severe with the bearings tested with the polyphenyl ethers, which were short lived.

Surface glazing, which can be present without spalling occurring, is illustrated in figure 6(b). Continued operation results in superficial pitting as shown in figure 6(c) and/or wear of the rolling-element surfaces. If operation of the bearing is continued under the condition shown in figure 6(c), spalling of the rolling-element surface will occur as is illustrated in figure 5. References 8 and 9 attribute the glazing phenomenon to marginal elastohydrodynamic film thicknesses. Under marginal elastohydrodynamic lubrication, high tangential forces can be induced which will tend to relocate the maximum shearing stresses closer to the surface (ref. 10). Under these conditions, more shallow fatigue spalls would be expected than normally occur under complete elastohydrodynamic conditions.

It is interesting to note that, with lubricant I (the synthetic paraffinic oil with the anti-wear additive), no glazing, superficial pitting, fatigue spalling, or wear occurred for all ten bearings tested. Each of these bearings was run for at least 180 hours. For the same lubricant without the additive (lubricant H) under the same conditions (i.e., temperatures between 550° to 600° F (560° to 588° K)), some glazing was observed. It is conceivable that anti-wear additives will reduce the tangential forces under marginal elastohydrodynamic conditions and thus increase bearing life.

Another mode of failure at elevated temperatures is that of "smearing." This mode is illustrated in figure 7. Smearing manifests itself by gross metal transfer, plastic

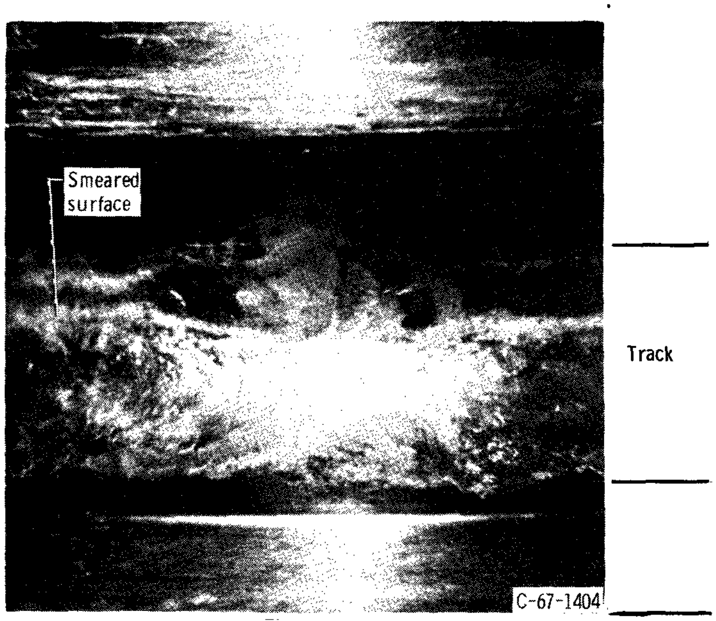


Figure 7. - Representative running track showing smeared surface.

deformation and/or galling of the rolling-element surfaces. It is believed that smearing occurs when the condition is primarily that of boundary lubrication accompanied by a high degree of sliding in the ball-race contact. This mode of failure was dominant for those bearings which failed other than by fatigue spalling.

Cage wear can be another failure mode in high-temperature bearing operation. Based upon work reported in references 6 and 11, cages were manufactured from AISI M-1 material of nominal hardness Rockwell C 58. To further assure minimal wear, the cages were silver plated. In all bearing tests in this investigation, cage wear was not a mode of failure.

Lubricant oxidation can be another mode of failure (ref. 1). When a lubricant starts to fail through oxidation, the result is usually formation of both soluble and insoluble compounds that may appear as resins, sludges, or acidic compounds. The resulting effects are the following:

(1) A gradual rise in viscosity of the lubricant

(2) The formation of adherent surface deposits that may interfere in small clearance spaces in the bearing

(3) Corrosion or deterioration of the metal parts

(4) General dirtying of the system by sludges or other insoluble compounds

Lubricant viscosity changes were not a problem in the tests reported herein inasmuch as fresh lubricant was periodically added to the test rig to replenish that lost because of evaporation or leakage. Consequently, bulk viscosity change was not apparent for the lubricants tested. In addition, corrosion or deterioration of the bearing elements did not occur or manifest itself with the various test lubricants. This may be due to relatively short testing times.

Formation of sludge appeared to be a minor problem with lubricant C, the 5P4E polyphenyl ether. However, this problem could be reduced either by the insertion of a copper screen in the lubricant test sump or by adding an oxidation inhibitor as in lubricant D. The experience with the other polyphenyl ether lubricants was similar. Sludging was not increased by exposing lubricant D to an air environment at elevated temperature. No deposit formations appeared on the bearings with the polyphenyl ethers.

The esters had a tendency to form hard coke deposits on the bearings at 500° F (533° K) under the low oxygen environment. The hydrocarbons exhibited no noticeably hard coke formations under the low oxygen environment, although the fluid became blackened and some sludge deposits were evident after extended running at 600° F (588° K).

Based upon the failure modes discussed herein, the lubricant temperature, stress, and life limits were estimated and are summarized in table IV. The bearing life potential was estimated based upon the lubricants performance relative to the appearance and performance of those bearings run with lubricant I, the synthetic paraffinic oil with the anti-wear additive.

Damage to the bearings in the form of surface distress and pitting of the race surfaces occurred in relatively short time periods with the polyphenyl ether lubricants. Consequently, the life potential with this family of lubricants was estimated to be only fair relative to the performance of lubricant I under the respective temperature and stress conditions listed. However, the bearings operated with lubricant C, the 5P4E polyphenyl ether without an oxidation inhibitor exhibited a poor life potential in the low oxygen environment even at lower stresses than that listed. The ester and hydrocarbon based lubricants showed good life potential relative to lubricant I with minimal surface distress at temperatures in the order of 500° and 550° F (533° and 560° K), respectively, under the limited oxygen environment.

SUMMARY OF RESULTS

Groups of 7205-size angular-contact ball bearings made from AISI M-1 steel were tested with eleven high-temperature lubricants. These lubricants were classified as polyphenyl ethers, esters, and hydrocarbons. Test conditions included outer-race temperatures of 250° to 600° F (394° to 588° K), speeds from 20 000 to 45 000 rpm, and maximum Hertz stresses of 189 000 to 347 000 psi (130 000 to 239 000 N/cm^2) in a SKF high-temperature bearing tester. The results of the tests were as follows:

1. Bearing AFBMA-rated (catalog) life can be obtained at a temperature between 550° and 600° F (560° and 588° K) with a synthetic paraffinic oil with an anti-wear additive under a low oxygen environment.

2. The estimated bearing life with a diester base lubricant was less than half bearing catalog expectations in the bearing temperature range of 450° to 500° F (505° to 533° K).

3. Of the bearings run with the three classes of lubricants, those run with the polyphenyl ethers exhibited the poorest life potential based on race groove surface distress. Early fatigue pitting, wear, and surface distress occurred in the rolling-element components.

4. Bearings run with the polyphenyl ethers have the ability to operate in an air environment although sludge formations occurred with this type lubricant in both air and low oxygen environments. Gross lubricant degradation did not occur with the hydrocarbon lubricants run in a low oxygen environment although the fluid became blackened and some sludge deposits were noticed. However, in the low oxygen environ-

ment heavy coke deposits formed on bearings run with the ester lubricants at a temperature of 500° F (533° K).

Lewis Research Center,
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Cleveland, Ohio, April 19, 1967,
720-03-01-01-22.

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